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Original Article

DETERMINATION OF STIFFNESS AND DAMPING COEFFICIENT OF TRACTOR FRONT TYRES IN NON-ROLLING CONDITIONS

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ABSTRACT

The suspension characteristics of agricultural tractors are dictated mainly by the characteristics of the tyres as they do not have spring suspension system like those provided on road vehicles. The tyre can be modeled approximately as a spring-damper system exhibiting under damped vibration.

The apparent spring stiffness and damping coefficient of a tractor front tyre was estimated in a non-rolling condition in the laboratory. For this purpose a test rig was fabricated. It had provision to measure the force transmitted by the wheel to a horizontal platform and to measure the deflection of axle. The wheel was dropped from a height of 50mm during the test. The force and deflection were recorded continuously using a data acquisition system at the rate of 33 sample readings per second per channel. The apparent spring stiffness and damping coefficient were estimated from the graph of displacement Vs time. The minimum and maximum values for stiffness are 156.73 kN/m and 174.196 kN/m respectively. The average value was obtained as 165.771 kN/m. And the minimum and maximum values for damping coefficient are 0.3752 kN.s/m and 0.5842 kN.s/m. The average value obtained as 0.46 kN.s/m.

***KEYWORDS:** Under Damped Vibration, Spring Constant, Damping Coefficient, Pneumatic Tyre and Drop test*

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INTRODUCTION

The tyre is the link between the road and the chassis of a vehicle. It has, therefore, an important role in the transmission of the tractive and braking forces and it influences the ride and the road stability of the vehicle (Laib, 1979). In certain cases the tyre influences the dynamic behavior of the vehicle significantly. The influence of the operational conditions on tyre characteristics is not well known.

Safe road handling and ride comfort depend on the dynamic behavior of on-road vehicles. If one assumes that road handling stability is a function of the tyre load and that acceleration is a measure of ride comfort then one can conclude that 10 percent stiffening of the tyre would reduce stability and comfort by approximately the same amount. Effective damping by the tyre is important for slow moving vehicles, which do not have a suspension system (example tractors). These vehicles have unfavorable dynamic properties, which is the result of inadequate tyre damping over and beyond the lack of suspension.

Tyres are the only suspension elements on most agricultural machines. The suspension characteristics of the tyres together with the masses and geometry of the vehicle and any machine or tool which is attached to it determine the ride vibration characteristics of the vehicle as it drives over rough ground surfaces (Lines and Murphy, 1991).

The vibration to which the driver is subjected has been reduced very little, where climatic protection, Roll over protection, noise reduction and improved controls have all made the tractor driver’s job easier and safer. Vibration causes long term injury to the driver and it generates high levels of stress in various parts of the machine. By reducing the vibrations of tractors the health and comfort of tractors drivers could be improved (Lines et al., 1989). Predicting the vehicle vibration is required in order to develop such vibration reducing measures effectively. Computer models that can predict the vibration of agricultural tractors have been produce by a number of authors; however, they are unable to predict the vibration with an acceptable degree of accuracy. The main reason for this inaccuracy appears to be a lack of information about the behavior of the tyres. The main objective of this study was the determination of deflection vs. load and time characteristics of an agricultural tractor tyre in non-rolling condition and estimation of stiffness and damping coefficient.

MATERIALS AND METHODS

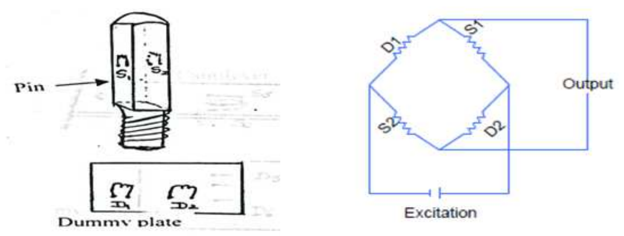
Measurement of Force Exerted By Wheel on the Platform

For measuring the force acting on the platform three transducer pins of rectangular cross-section were arranged under the platform. These were fitted to the stand with nut and bolt arrangement. The platform was freely supported on the three pins. One pin is on one side and another two pins are on the opposite side. The force experienced on these two sides were measured separately and added to get total force acting on the platform. The pins are subjected to compressive forces. Two strain gauges were fixed on one pin as shown in Figure 1.a. These are active gauges (S_1 and S_2). These two active gauges were connected with another two dummy gauges (D_1 and D_2) to form a Wheatstone bridge circuit as shown in figure.

On the other two pins, the strain gauges were fixed with one strain gauge on each pin. Strain gauges S_3 and S_4 are shown in Figure 1.b. These two active gauges were connected with another two dummy gauges (D_3 and D_4) to form a Wheatstone bridge as shown in figure. The overall dimensions of the pins were 1.5 × 1.5 × 10 cm.

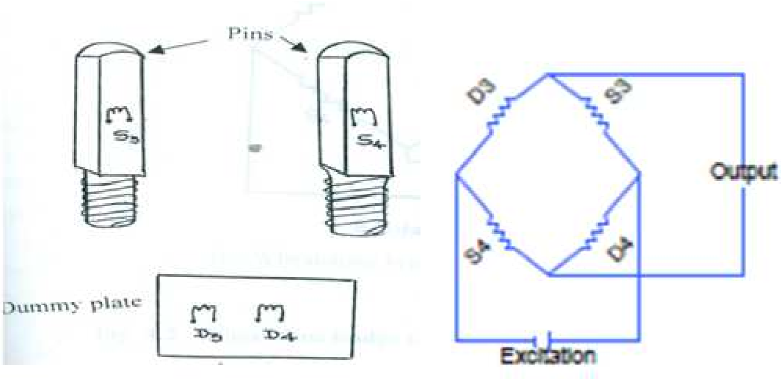
Measurement of Deflection

For measurement of deflection, cantilever beam was used. A 36 cm long and 2.5 cm wide cantilever bar with 1 mm thickness was used Figure 2. At the middle of the bar two strain gauges (S_5 and S_6) were fixed on the upper and lower sides of the beam. The upper one is subjected to tension and bottom one is subjected to compression. These two are connected with another two dummy gauges (D_5 and D_6) to form a Wheatstone bridge Figure 2.



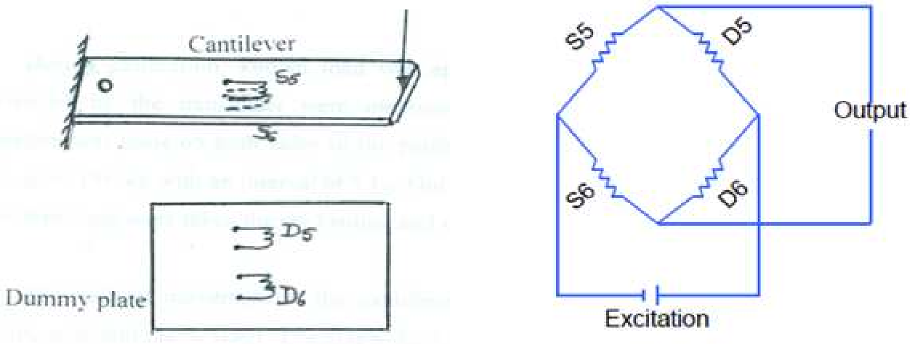
a. Arrangement Of Strain Gauges B. Wheatstone Bridge Circuit

a. One Pin Side



b. Two Pin Side

Figure 1: Wheatstone Bridge Circuit for Measurement of Force



(a) Arrangement of Strain Gauges (b) Wheatstone Bridge Circuit

Figure 2: Wheatstone bridge Circuit for Measurement of Deflection

The cantilever beam was fixed to a 1.5 m high stand so that the free end of cantilever beam moves up and down freely. One can fix the beam at a desired height. A 2.5 cm diameter hollow GI pipe with a circular base was used as stand. The circular base will give stability to the stand. The stand and cantilever arrangement are shown Figure 3.

During calibration, known load was applied on the platform and the strain experienced by the transducer was measured with a Wheatstone bridge circuit. Calibration was done on both sides of the platform separately. The dead weights were added up to 150 kg with an interval of 5kg. Output was measured with the help of data logger. Readings were taken during loading and unloading.

The vertical movement of the cantilever bar was measured with the help of hydraulic jack and graph sheet. The graph sheet was fixed to the vertical wall. With the help of hydraulic jack the cantilever bar was slowly loaded and corresponding vertical deflection was measured on graph sheet upto 15 cm with the interval of 1 mm. The corresponding output voltage was noted with the help of data logger. This was done while loading and unloading of the cantilever bar.

Stand and Platform

A stand was fabricated using a 60 × 60 × 7 mm ms angle iron to support the wheel and help in measurement of forces and deflection was shown in Figure 4. The overall size of the stand is 63 × 30 × 60 cm . A 5 mm thick MS sheet of 30 × 60 cm was placed on the top of the stand, which is supported on three square cross-section pins. These three pins are attached to the stand with nut and bolt arrangement. The three supporting pins are made of 1.5 × 1.5 × 10 cm, and strain

gauges were mounted to measure the force. In order to restrict the rolling of wheel the stand was provided with two supports on either side of the stand. The supports were erected to a height upto 70 cm from platform.

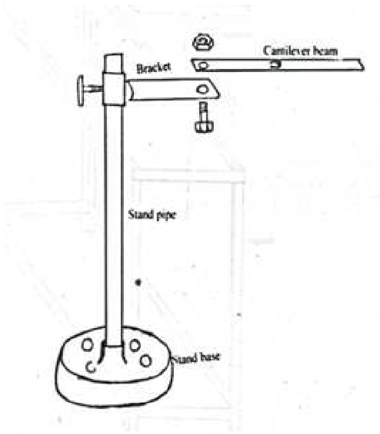


Figure 3: Stand and Cantilever Beam for Platform Measurement of Deflection

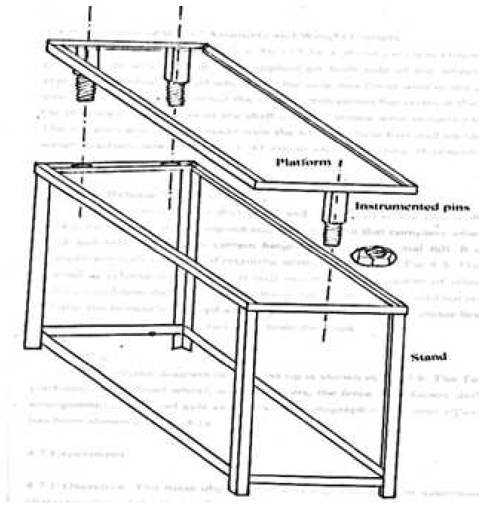


Figure 4: Schematic Diagram of Stand and Platform Measurement of Deflection

Preparation of Wheel Assembly and Weight Carriers

The tractor front tyre 6 × 16″ (15.24 × 40.64 cm) was fastened with a shaft 3.34 cm diameter with a shaft 3.34 cm diameter to carry the loads applied on both sides of the wheel. A guide disc was provided on either side of wheel and the axle was fitted with in the guide pillers. So the guide discs does not obstruct the vertical movement but restricts the tyre lateral turning. On the extended portion of the shaft weight carriers were mounted to fasten the weight. The weight carriers were made with the help of angle iron and as ms sheet. The size of the weight carriers was kept 43 × 43 cm to enable loading of concentric discs of higher weight.

Wheel Release Mechanism

In order to evaluate the forces and deflection of the tyre with respect to time, a release mechanism was designed and fabricated so that complete wheel assembly can be lifted and released from a certain height for a gravitational fall. It consists of a lever arrangement with the help of retaining springs as shown in Figure 5. The release bracket is pivoted at release lever pivot. In loading condition the hook touches the roller. So the load should not release. If we insert rod into the lever pipe and press downward then the release bracket will move upward then the load will fall suddenly from the hook.

Test Rig

Test rig (Figure 6) consists of a platform, tractor front wheel, weight carriers, the force transducers, deflection measuring arrangement and wheel axle assembly.

Procedures

The experimental setup was prepared by placing the stand over a horizontal ground and it should be convenient to lift the wheel with the help of crane. Three force transducer pins were fixed to the with nut and bolt arrangement. The MS platform was mounted on these pins and wheel axle assembly was kept on the platform such a way that the axles was fitted in the gap of two vertical supports and tighten the bush nuts to restrict the side movement of the tyre. The weight carriers

were mounted on the both side of shaft carrying the wheel. It was assumed that the distance of weight carriers on either side remains same. 62.5 kg weights were added on each side and tyre pressure was maintained at 1.75 kg/cm^2. The deflection measurement cantilever bar was mounted on the adjustable stand to adjust the height of the cantilever bar. The cantilever ar was kept close to the wheel axle and cantilever bar was adjusted such that it simply touches the wheel shaft. The 12V excitation was given for three bridges and output terminals were connected to the input ports of the data logger. All the three wheat stone bridges were balanced using excitation null balance system. The complete wheel assembly was lifted to a height of 5 cm above the platform. The release mechanism was fitted in between wheel and crane hook. The release mechanism was operated so the complete assembly would fall freely and hit the platform. The data on load and deflection were recorded at an interval of 0.01 seconds. The data was transferred from data logger to computer into MS-Excel package with the help of HP interface software. After application of calibration equations to the original data, the plots were drawn with respect to time vs deflection and time vs load. With the help of these plots the characteristics of tyre (stiffness and damping coefficient) are calculated. The test was repeated 15 times.

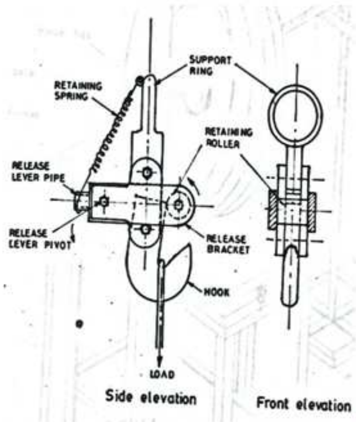


Figure 5: Release Mechanism for Dropping the Tyre from a Height

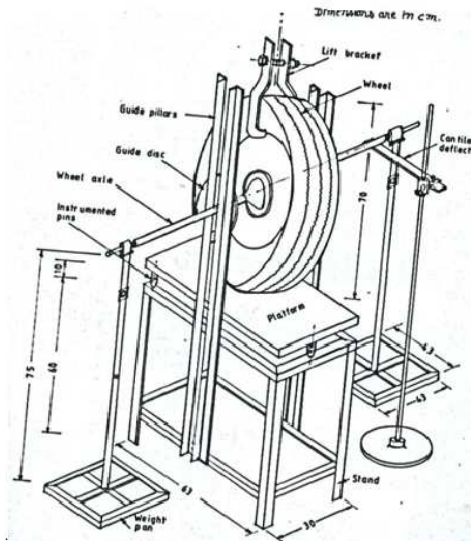


Figure 6: Laboratory Setup for Determination of Non-Rolling Deflection Characteristics of Tyre

RESULTS AND DISCUSSIONS

Calibration of Instrumented Pins

- **Single Pin Side**

Calibration was done with the help of dead weights, data logger. The readings were taken while loading and unloading. The average values at the loading and unloading was calculated and a best fit relationship was drawn between load applied on the platform (kg) and output voltage (mV). The best fit equation obtained is as follows

$$Y = 449.43X - 7.9892 \qquad (R^2 = 0.9999) \tag{1}$$

Where,

Y = load on the one pin side, kg

X = output voltage, mV

R^2= Coefficient of Correlation

- **Two Pin Side**

Calibration was done with the help of dead weights, data logger. The average values of output voltages at loading and unloading were calculated. A plot between output in mV and applied load on the platform was drawn. Output was taken on X-axis and load on the Y-axis. The values are in linear relationship. If we increase the load the output voltage also increases. The best fit relationship equation was as given below

$$Y = 497.5X - 0.014 \qquad (R^2 = 0.9992) \tag{2}$$

Where,

Y = load on the two pin side, kg

X = output voltage, mV

R^2= Coefficient of Correlation

Calibration of Cantilever Beam

While increasing the load on the cantilever bar the vertical deflection also increases. The deflection and output voltage are in linear relationship. The average value of output was taken while loading and unloading conditions. A graph was drawn between deflection (mm) and output voltage (mV) shown in. The best-fit equation obtained is

$$Y = 22.826X - 0.37 \qquad (R^2 = 0.9994) \tag{3}$$

Where,

Y = Deflection, mm

X = output voltage, mV

R^2= Coefficient of Correlation

Force and Deflection Characteristics

The deflection vs time and force vs time plots were drawn shown in Figure 7 and Figure 8, respectively. The force is decaying with respect to time. In both cases, the amplitude decays over time. The deflection vs time graph shows that the vibration is in an under damped condition.

Stiffness and Damping Coefficient

Stiffness and damping coefficient are calculated from the amplitude decay curve obtained from drop test. The test was replicated 14 times. Each time the stiffness and damping coefficient was calculated and are shown in Table 1. First the smooth envelopes were drawn to the amplitude decay curve. The logarithmic increment was calculated Eq (4) and after that the damped natural frequency of the decay curve calculated by using Eq (5). By substituting the above obtained values in Eq (6) and Eq (7) damping coefficient and stiffness can be calculated.

$$b = 2\sqrt{D_p\,\delta - \delta^2} \quad (4)$$

$$I = c_d\sqrt{d\delta - \delta^2} \quad (5)$$

Where, c_d = Coefficient

D_p = A function of tyre width and height of tyre cross section and

δ = Vertical tyre deflection

$$\frac{f^2}{w} = c_1 + \frac{c_2 f}{(p_i + p_c)} \quad (6)$$

$$\frac{f^2}{w} = 0.0838 + \frac{5.9559f}{(p_g)} \quad (7)$$

Where, f = Deflection of tyre, mm

W = Normal load, daN

P_i = Inflation pressure, kpa

c_1 and c_2 are constants

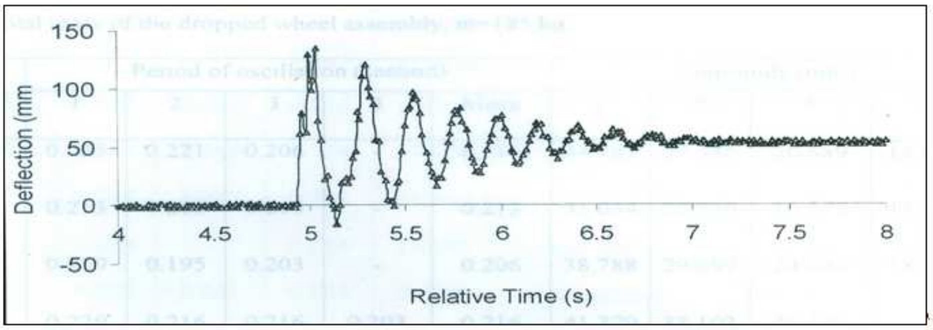


Figure 7: Deflection Vs Time Plot

The minimum and maximum values for stiffness are 156.730 kN/m and 174.196 kN/m, respectively. The average value was obtained as 165.771 kN/m. And the minimum and maximum values for damping coefficient are 0.3752 kN.s/m and 0.5842 kN.s/m. The average value is obtained as 0.46 kN.s/m.

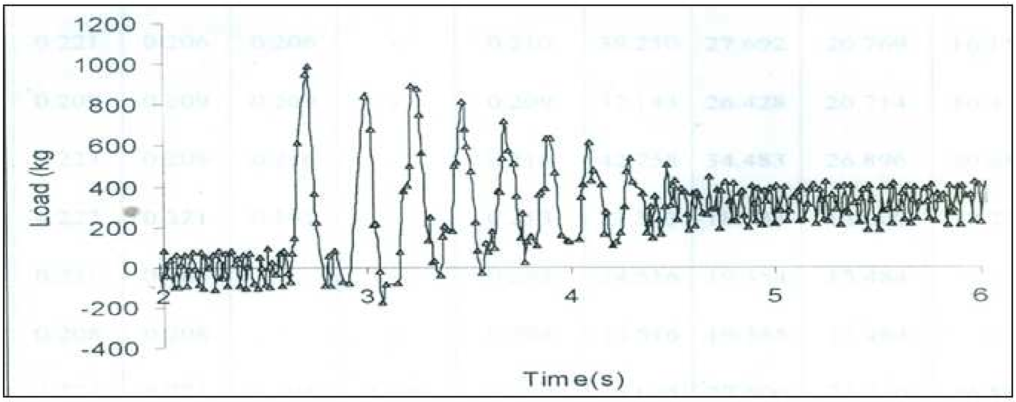


Figure 8: Load Vs Time Plot

Table 1: Observed Values of Period of Oscillation and Amplitude

Observation No	Period of Oscillation (Second)					Amplitude (Mm)			
	1	2	3	4	Mean	1	2	3	4
1	0.205	0.221	0.206	-	0.211	34.483	27.586	20.689	15.862
2	0.213	0.213	0.213	-	0.213	31.034	22.759	16.552	11.724
3	0.219	0.195	0.203	-	0.206	38.788	29.697	24.242	18.182
4	0.229	0.216	0.216	0.203	0.216	41.379	33.103	26.307	-
5	0.235	0.221	0.206	0.206	0.216	34.483	25.517	19.310	15.172
6	0.207	0.207	0.200	-	0.205	27.407	20.740	14.815	-
7	0.221	0.206	0.206	-	0.210	39.230	27.692	20.769	16.153
8	0.209	0.209	0.209	-	0.209	32.143	26.428	20.714	16.428
9	0.223	0.209	0.216	-	0.216	42.758	34.483	26.896	20.689
10	0.227	0.221	0.191	-	0.213	23.333	18.333	15.000	12.222
11	0.231	0.185	-	-	0.208	24.516	19.354	15.484	-
12	0.208	0.208	-	-	0.208	24.516	19.355	15.484	-
13	0.221	0.221	0.191	0.206	0.209	35.625	27.500	21.250	16.800
14	0.207	0.219	0.201	-	0.209	37.241	26.206	19.310	13.793

Total mass of the dropped wheel assembly, m = 185 kg

Table 2: Calculation of Stiffness and Damping Coefficient of Tyre

Logarithmic decrement (Mean)	$\omega_d = 2\pi/\tau$	$a = 4m^2\omega_d^2\delta^2$	$b = \delta^2 + 4\pi^2$	$c = \delta^2/(\delta^2 + 4\pi^2)$	Damping coefficient (kN.s/m) $C_{eq} =$	Stiffness, $k_z = m\omega_d^2$ / 1-c (kN/m)
0.258	29.778	8080419.6	39.545	0.00168	0.452	164.274
0.324	29.498	1250485.4	39.583	0.00265	0.563	161.402
0.253	30.541	8173555.1	39.748	0.00162	0.455	172.849
0.229	29.088	6047885.8	39.531	0.00132	0.402	156.73
0.274	29.088	8696255.6	39.553	0.00189	0.469	156.83
0.307	30.649	12120285	39.573	0.00238	0.554	174.196
0.296	29.92	10737670	39.566	0.00221	0.522	165.981
0.224	30.06	6206938.4	39.529	0.00126	0.397	167.379
0.242	29.088	6783626.1	39.537	0.001487	0.414	156.763
0.216	29.942	5557712.7	39.525	0.00118	0.375	161.161

Table 2: Contd.,

0.229	30.207	6550728.6	39.531	0.00132	0.407	169.029
0.229	30.207	6550728.6	39.531	0.00132	0.407	169.029
0.249	30.063	7671261.3	39.54	0.00157	0.441	167.4626
0.33	30.063	13473982	39.587	0.00275	0.584	167.661

Average of Damped natural frequency, ω_d = 418.636/14 = 29.903 rad/s

Average of logarithmic decrement, δ = 3.66/14 = 0.261

Average of stiffness, k_z = 2320.7856/14 = 165.770 kN/m

Average of Damping coefficient, c_{eq} = 6.44221/14 = 0.460 kN-s/m

CONCLUSIONS

The wheel exhibited under damped vibration during drop test. The minimum and maximum values for stiffness are 156.73 kN/m and 174.196 kN/m, respectively. The average values was obtained as 165.771 kN/m. And the minimum and maximum values for damping coefficient are 0.3752 kn.s/m and 0.5842 kN.s/m. The average value obtained as 0.46 kN.s/m. The frequency of vertical oscillation was found to be 4.747 cycles/s. Time constant for decay of amplitude was found to be 0.79 seconds.

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